REVAMP OF TURBOCOMPRESSERS

by

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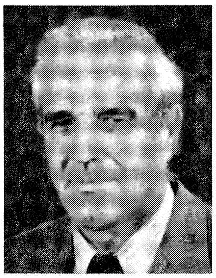

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ABSTRACT

The rerating possibilities of compressors are discussed. Starting from the characteristic equations for design purposes, the relevant design parameters are specified. Concerning the rerating of axial compressors, methods are suggested for the rerating of axial compressors and for centrifugal compressors. Examples of revamped compressors are presented.

INTRODUCTION

Turbocompressors are the most vital components in any process plant where gases have to be transported or must be compressed to process requirements. Turbomachinery and its drivers are designed and manufactured to last decades of uninterrupted operation and if properly maintained they do so.

During the planned and expected lifetime of the turbo group technology of the production process, requirements of gas pressure and mass flow, gas composition, etc., very probably will change. Hand in hand with the adjustments and reworking of the various process installations, quite often a rerating of the compressor train is inevitable, in order to obtain the stepped up production goals, or to avoid an upset of the economics of a reduced plant output. In any case, process compressors are engineered and tailored to meet the exact specified gas flow at the required pressure and temperature of the various operating points. Therefore, in the design stage, the mechanical and aerodynamical designer already have to demonstrate flexibility to fit mostly standardized components into a machine which fulfills the expectations of the client. Thus turbocompressors inherently are designs which can be modified with reasonable effort.

THEORETICAL BACKGROUND

Aerodynamic Lay Out

Necessary changes in the behavior of a compressor executed can best be demonstrated in briefly explaining the aerodynamic design of an axial compressor. The overall performance is derived by the stage stacking method, which allows treating each individual stage in a straightforward manner. The basic design tools are stage characteristics, which show the head factor and the efficiency of a stage in function of a flow factor (Figure 1). A stage comprises a row of rotor blades and a row of stator vanes (respectively, an impeller and a diffuser in a centrifugal compressor). Choosing the flow factor pinpoints the design point of a particular stage and knowing the suction volume flow the necessary area of the flowchannel can be calculated:

$$A = \frac{\dot{V}_1}{\phi \cdot u_2} \qquad (m^2) \tag{1}$$

The dimensionless headfactor allows the computation of the produced head and pressure ratio:

$$hp = \mu p \cdot u_2^2 \qquad (j/kg) \tag{2}$$

$$hp = T_1 \cdot R \cdot Z_1 \cdot \frac{n}{(n-1)} \left[\pi^{\frac{n-1}{n}} - 1 \right] \qquad (J/kg) \tag{2a}$$

$$\pi = \left[\frac{\mu p \cdot u_2^2 \, (n-1)}{T_1 \cdot R \cdot Z_1 \cdot n} + 1 \right]^{\frac{n}{n-1}} \qquad (-) \tag{3}$$

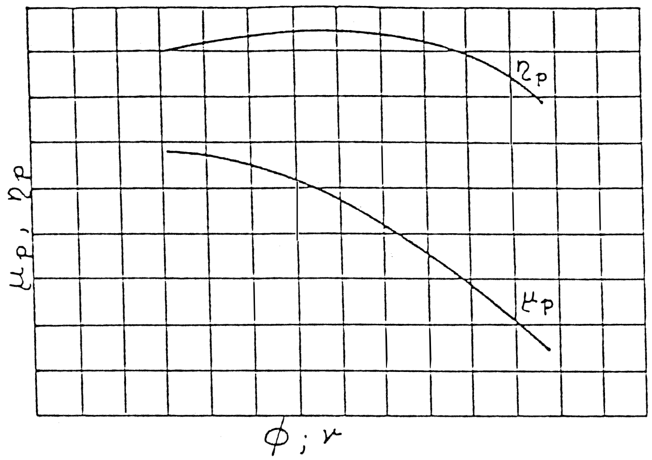

Figure 1. Stage Charcteristic.

Taking the polytropic efficiency from the stagecharacteristic the discharge temperature and gaspower can be determined:

$$T_2 = T_1 \cdot \pi^m \qquad (°K) \qquad (4)$$

$$P_i = \frac{\dot{m} \cdot hp}{\eta p} \qquad (W) \qquad (5)$$

The pertinent thermodynamical data of the stage are thus known. The density change across this stage determines the suction-volumeflow of the downstream stage which can now be designed in the prescribed manner.

$$\frac{\rho_2}{\rho_1} = \left[\frac{\mu p \cdot u_2^2\,(n-1)}{T_1 \cdot R \cdot Z_1 \cdot n} + 1\right]^{\frac{1}{n-1}} \qquad (-) \qquad (6)$$

The overall performance is obtained by adding up headrise and internal power of the individual stages.

Axial Compressors

In contrast to gas turbine compressors, industrial type compressors have to be designed for a wide and flexible operating range, and must be equipped with a means for controlling the performance. If the driver has variable speed the problem usually is solved. For constant speed drive, variable stator vanes are placed in the front stages, or in all stages if a wide operating range is demanded.

By varying the pitch of the stator vanes, the direction of the absolute velocity approaching the downstream rotor blades varies more or less with the change of the geometrical angle; simultaneously, the mass flow and, consequently, the meridional velocity decreases (closing of vanes) or increases with the result that the incidence angle of the relative velocity at the inlet of the rotor blade varies only slightly. This fact assures stable flow conditions and a constant level of efficiency over a wide swing of stator vane settings (Figure 2).

The performance of stage characteristics can reasonably well be predicted analytically, but the final design documents are still obtained from tests of research compressors. Despite powerful computer programs, the influence of Mach and Reynolds number, of blade tip clearance and blade geometry can more reliably be established experimentally. This information is needed not only for the initial dimensioning, but is also for predicting a revamp of the compressor. The selection of the setting angles of the individual stages at the design point and the choice of the flow channel area distribution leaves two degrees of freedom for an optimum layout. Furthermore the adjusting mechanism of the stator vanes permits a descrete angular vane movement at each row, offering the possibility to place off design operating points at favorable efficiency range within the pressure volume characteristic.

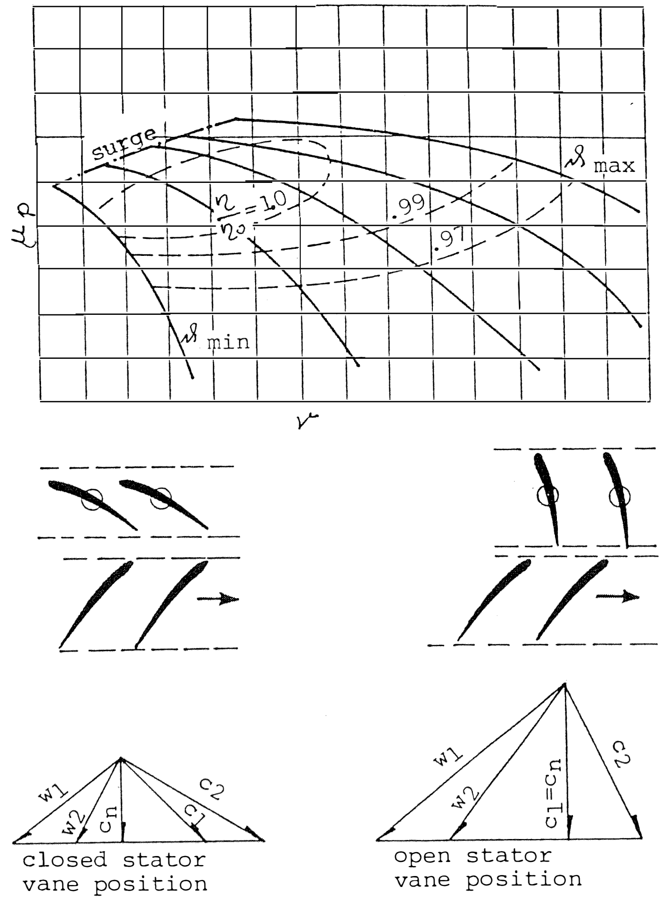

Figure 2. Stage Characteristic of an Axial Compresor with Variable Stator Vane Settings.

The hub velocity of a 50 percent to 80 percent reaction blading is some 185 to 165 m/s resulting in a polytropic head increase per stage of approximately 16000 to 17000 J/kg (= 5350 to 5685 ftlb/lb). In an air compressor, this head is equivalent to a pressure ratio of approximately 1.25 in the first stage, and due to the temperature rise of the gas, approximately 1.12 in the last stage, if the unit is designed for an overall pressure ratio of 5. Under these conditions, the density ratio across the first stage is 1.12 to 1.15 and, as already outlined, the area of the flow channel will decrease by the same ratio. By removing the first stage, the suction volume flow consequently would be reduced by 12 percent to 15 percent, and at the same time, the overall polytropic head would drop by the head contribution of this stage. If space is provided, the loss of pressure ratio could be compensated by installing an additional stage at the rear end of the rotor. The area reduction of the flow channel in a first approximation is proportional to the density increase of the gas. An indication how many stages would have to be removed to end up at the required down rating is given in Equation (6).

Consequently, head reduction at unchanged mass flow can be achieved, if one or more rear stages are removed. By this staightforward aerodynamics, and the fact that axial compressors usually comprise many stages, a fine tuning of the performance can be accomplished. Possible performance variations by restaging are presented in Figure 3.

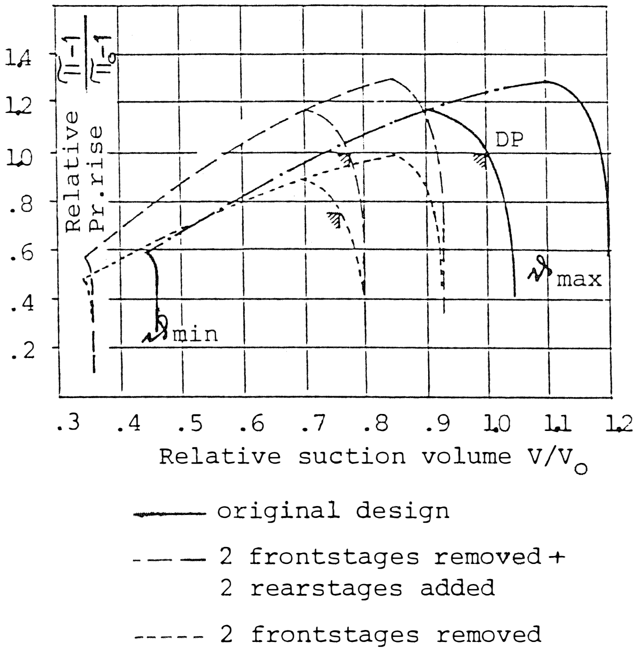

Figure 3. Performance Variation of an Axial Compressor by Restaging.

A further procedure for minor and well predictable performance adjustments presents itself if the explained stage by stage design procedure is applied. With reasonable effort, the schedule of the stator vane settings can be altered by exchanging the levers of the adjusting mechanism. This opens the possibilty to move, for example, the island of best efficiency in the region of the newly defined operating points, to extend the characteristic towards smaller flow or to increase the maximum flow, if the limits of Mach number and stone wall are not yet reached. For major adaptions to a newly specified flow range, there always exists the possibility of replacing the blade carrier, combined with a modification of the blades. Lower flow demand causes no problem. The new blade carrier will be designed for the new specification, and very probably the blades of all stages can be reused after being machined to the dimensions of the new flow channel.

The downrating of a blast furnace blower by installing of a new blade carrier and by modifying the schedule of the stator vane setting is described under *Down Rating and Moderinization of Furnace Blowers*.

Centrifugal Compressors

Impeller Modification

Rerating of centrifugal compressors follows the same aerodynamic laws as outlined for axials. Shifting around impellers, however, is not so easily accomplished, and only big steps in performance change can be realized. (One centrifugal stage produces approximately the same head as three axial stages).

Instead of removing the first impeller, for example, it could be replaced with one of lower or higher flow capacity. In order not to disturb the matching of the remaining stages, this new impeller, in conjunction with an adapted diffuser, should produce a corresponding pressure rise which renders an unchanged suction volume flow to the downstream impeller. The effect of this measure upon the overall characteristic is identical to the one obtained by varying the mechanical speed. Fine tuning of pressure and flow reduction can be realized by trimming the first and possibly the second impeller. By trimming the outer diameter of the blades, the diameter, but not the cover disk is reduced, so the shroud and disk do not have to be machined. The thus formed rotational diffuser is necessary for a proper transport of the flow between impeller and diffuser. Normally, the diffuser must not be replaced. From the velocity triangles at original and cutback conditions, it can be approximated that the work input of the impeller is proportional to the square of the tip speed u_2 i.e., proportional to the square of the tip diameter of the blade (Figure 4). (The blade angle β_2 does not vary much at the outer part of the blading.)

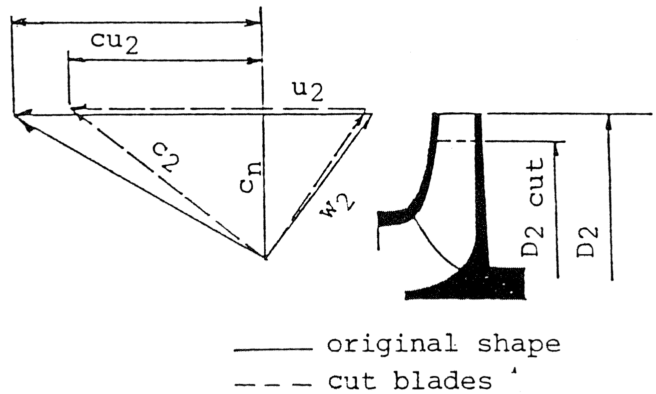

Figure 4. Exit Velocity Triangles Cut and Uncut for Impeller Blades.

In contrast to the possible prediction of μ_o-reduction, the accompanied efficiency decrease cannot so readily be established. It depends upon the basic aerodynamic design, the specific speed, matching between diffuser and impeller, and exact corrections can only be obtained by tests. In most aerodynamic designs of moderate Mach numbers, the diffuser determines stall. From the velocity triangles, it can also be noticed that the flow angle of the absolute velocity (C_2) approaching the the diffuser becomes steeper and the critical flow angle at which stall is triggered is shifted towards smaller flow.

Diffuser modification

The presented stage characteristic is a combination of the performance of impeller and diffuser. The impeller converts only a part of the work input into static pressure rise, the remaining kinetic energy is partly converted into potential energy in the diffuser. The split of the performance can be pinpointed, if in the course of testing a stage, pressure and temperature at the exit of the impeller are also recorded. Generally, the characteristic of the impeller has a wider volume range between stall and choke than the diffuser characteristic. Therefore, the performance of the diffuser, bladed or unbladed, limits the range of the overall stage characteristic. This fact offers the further possibility to influence the behavior of the stage by altering the geometry of the diffuser (Figure 5). In contrast to the bladed diffuser, performance adjustment via an unbladed diffuser, is very limited. Thus, the following explanation is valid only for bladed diffusers. As mentioned previously, with decreasing mass flow, the angle of the flow approaching the diffuser is reduced, and once this angle is markedly smaller than the geometrical angle of the leading edge, the flow will separate and trigger stall. The maximum flow capacity is reached when the narrowest flow area within the diffuser channel chokes the flow.

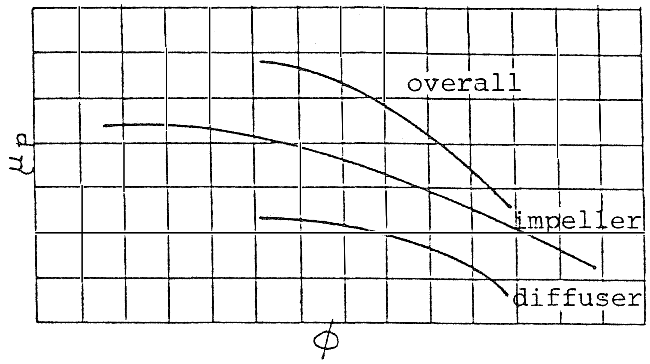

Figure 5. Split of Performance between Impeller and Diffusor.

If the volume flow of a stage must be reduced, and especially if the surge point needs to be shifted towards smaller flow, a new diffuser with a smaller blade inlet angle will render the desired result. A similar performance correction can be achieved by narrowing the flow path of the diffuser channel. The increased meridional velocity lifts the flow angle and the critical minimum angle is reached at a reduced mass flow. If the diffuser vanes are cantileverd, this would be the less expensive alternate, since the blades of the existing diffuser can be machined to the necessary height. An additional prerequisite would be the possibility to shift the cover disk of the blades in order to fill the created axial gap.

Increase of Rotational Speed

Rerating by adjusting the rotational speed is a very predictable and an economical way, if the driver and possibly the gear rating allow this undertaking. However, the mechanical and aerodynamical design like critical speeds, rating of the bearings, axial thrust, blade vibrations, Mach number, etc., must be scrutinized. Usually, the plant resistance line moves along a parabola, similar to the way the region of best efficiency is displaced within the extended volume pressure ratio characteristic, and, therefore, the specific power consumption will not be affected. Under *Uprating of an Oxygen Compressor Set*, rerating of oxygen compressors via speed increase is presented.

Improved Aerodynamic Efficiency

Research to improve the aerodynamic efficiency must be a continuous and essential ambition. The compressor installation described in *Efficiency Improvement* was revamped purely to improve the power consumption. Mass flow and head rise remained unchanged. New impellers, especially in the range of low specific speed, and also newly designed diffusers and return channels, lowered the electrical power input by some nine percent.

EXAMPLES OF EXECUTED REVAMPS

Down Rating and Modernization of Blast Furnace Blowers

Plant Description

A steel work in Spain operates two blast furnaces, each fed by one motor-driven axial compressor—a third compressor serves as standby. These blowers were commissioned in 1970–71 (Figure 6).

A complete modernization of one blast furnace was envisaged for the reline in 1986. Discussions started in 1984 on how to adapt the blowers to their new duties:

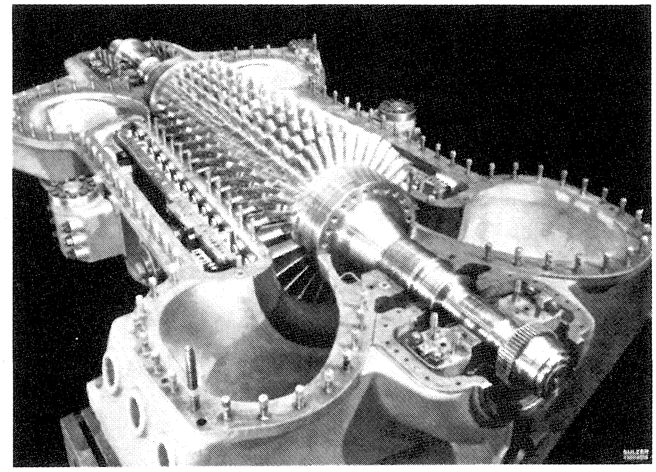

Figure 6. Blast Furnace Blower.

- Capacity should be reduced by 35 percent to 40 percent at constant head.
- The existing pneumatic/hydraulic control system was considered inadequate in accuracy and response. On top of that, costs for spares were considered very high, and time spent for servicing the old loops was longer and longer.

As investigations went on, the two following solutions were proposed and accepted by the client.

Blower Modification

Original and new performance curves are shown on Figure 7. This reduction in flow was obtained by the following measures:

- Shorten the blade height by about six percent. The original blades were reused on the rotor and stator. The blade carrier was replaced to match the smaller blade tip diameter.
- Stagger angle for the guide vanes is improved. Advantages: Flow control for the original design was with guide vane setting from 50 degrees to 90 degrees. The new staggering of the guide vanes allows 23 degrees to 90 degrees. Suction throttling for startup and the lower end of the performance map is no longer required.

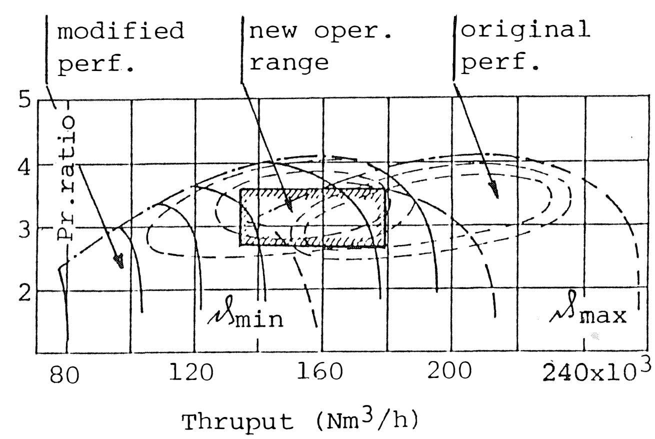

Figure 7. Performance Map of Blast Furnace Blower Before and After Downrating.

Control System Replacement

The three compressors were equipped with a hydraulic pneumatic control system for:

- flow control

• surge prevention control system

• pressure limitation.

Today's digital electronic control system fulfills the same duties of course, with the advantages of:

• digital control systems are built with standardized elements; they are tailored to customer's requirements with the corresponding software.

• quick response and good accuracy. Variations within the cold blast system or in the blast furnace are measured quickly and corrective measures are taken. As a result, only 50 percent of the original safety margin between surge limit and blowoff line is needed. Since settings in the controllers and transmitters will not shift in time, and because of their excellent accuracy, the adjusted safety margin remains very stable.

Also the cold blast flow control takes advantage of the accuracy of the new control system.

• new control system provides advantages not only for the operation, but for maintenance as well. The modular design allows checking their function very easily and replacibg faulty parts, if necessary.

Conclusion

The complete down rating and modernization on all three compressors was realized in 1986, keeping one compressor online and a second one on standby. An acceptance test demonstrated that the new designed compressors fulfilled the specification.

Minor modifications were required on the power supply loops of the control cubicles. After the corresponding parts were replaced, the customers staff was easily familiarized with the new systems.

These modernized compressor sets combine proved installation with the most modern technology in compressor design and control, fit for many years of troublefree operation.

Uprating of an Oxygen Compressor Set

Plant Description

The oxygen for an Austrian steel mill was supplied by different air separation units, working in parallel on a common header. More and more problems were caused by a very unstable oxygen demand and pressure. The decision was made to run the air separation units at a constant production and to feed oxygen to an oxygen tank. The pressure of this tank was floating and could reach 25 bar, compared with 16.7 bar which is the design pressure for the turbocompressor in question. This unit, previously providing the baseload, was becoming useless, because of its very low discharge pressure.

Uprating Study

The compressor set is a motor driven, two casing compressor. The investigations showed that:

• pressure test of the HP casing was at 36.8 bar. Operating pressure of 24.5 bar was, therefore, the target for the entire uprating.

• circumferential speed of the biggest impeller had some margin for a speed increase.

• lateral critical speeds were calculated at 4,500 rpm (LP rotor) 4,700 rpm (HP rotor) and 13,500 rpm (spacer between low power (LP) and high pressure (HP) rotors).

• motor and gearbox were able to cope with a higher duty.

• coolers and pipes were not designed for this higher pressure and had to be replaced.

Modification Required

Speed Increase. A careful investigation showed that a speed increase from 11,833 to 12,444 rpm was acceptable for all impellers. This new speed results in a surge pressure of 29 bar, enough safety margin to cope with the 25.5 bar required. Original and new performance is shown on Figure 8.

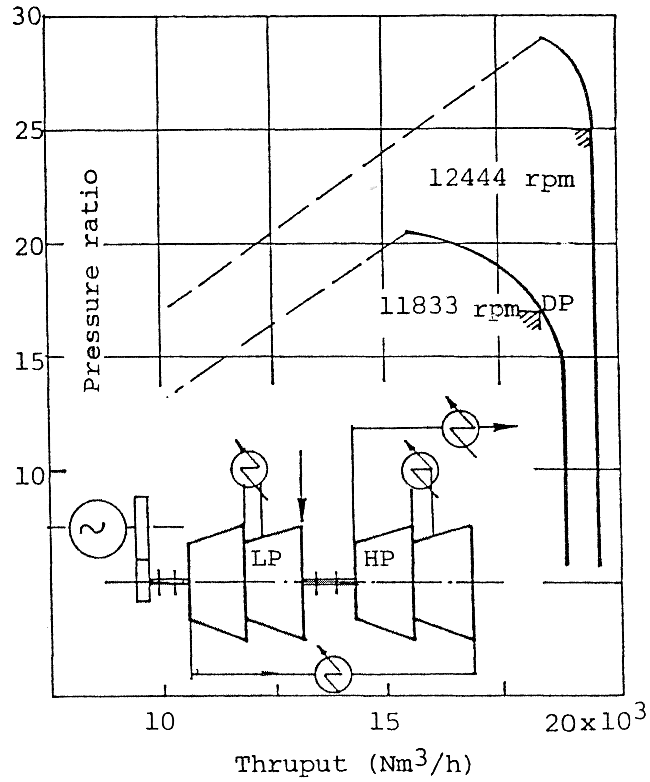

Figure 8. Performance Improvement by Speed Increase.

A gear set for the new speed and performance was designed. The existing motor was able to cope with the power increase from 2,310 kW to 3,060 kW.

Coolers and Piping. New data for each nozzle were given to the customer, and also allowable forces and moments. The customer's engineering division was then busy checking and redesigning the entire system.

Critical Speed. There is a 8.5 percent safety margin between the operating speed of 12,444 rpm and the critical speed of 13,500 rpm. Tilting pad bearings were installed to ensure a sound mechanical performance of the uprated train.

Miscellaneous. Axial thrust at the increased discharge pressure was still within the limits of the axial bearing. Some valves with actuators had to be replaced due to the higher pressure. The lube oil pump was replaced to deliver the higher oil flow required by tilting pad bearings. The main motor is started with reduced voltage. This remains true also for the new compressor speed.

Conclusions

The modified compressor was commissioned in 1983. It now delivers eight percent more oxygen at a 40 percent higher delivery pressure. The compressor is running smoothly with low and stable vibrations since its modification.

Efficiency Improvement

Plant Description

In an air separation plant in France a centrifugal compressor serves as a nitrogen recycle compressor. The plant was ordered in 1977, and commissioned in 1979. The customer was very satisfied with this compressor. Discussions started in 1984, however, to reduce its power consumption, which seemed to be high when compared with newer plants.

The compressor comprises six stages with intercooling, driven by an electric motor (Figure 9). Its performance perfectly met the specifications, so that exactly the same flow and pressure were aimed for, but with a lower power consumption after modification.

		before	after modification
Flow	kg/s	16.22	16.22
Suction pressure	bar	5.9	5.9
Temperature	C	19	19
Discharge pressure	bar	35.4	35.4
Speed	rpm	11,400	17,400
power at motor coupling	kW	4,220	3,886

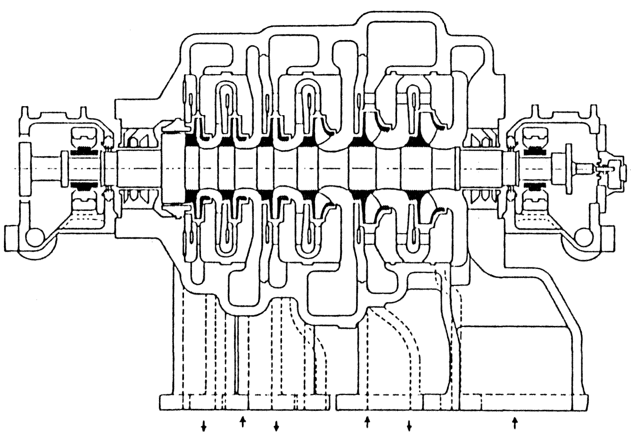

Figure 9. Nitrogen Compressor.

New Design

The customer required in an early stage that existing parts should not be modified. They liked to use them as emergency spares. It was obvious, therefore, that the rotor and all internals had to be replaced. A gain in efficiency of eight percent was guaranteed. Efficiency improvement per stage is reflected in Figure 10.

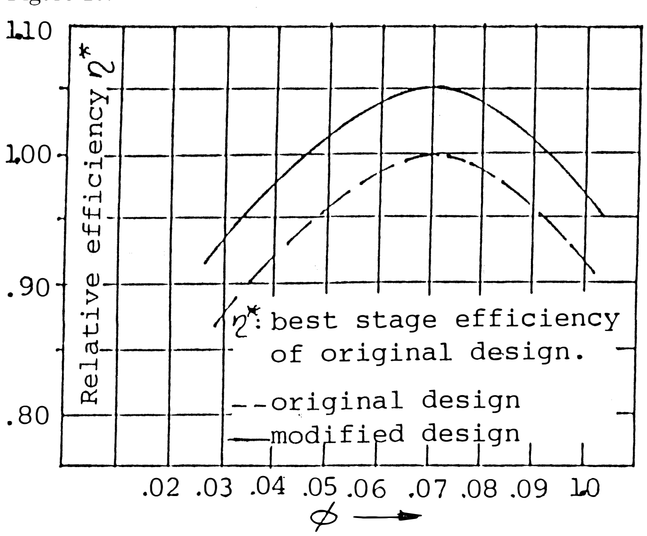

Figure 10. Comparison of Shape Efficiencies Between Original and Uprated Design of Axial LP Compressor.

Result

The new parts for the compressor were installed in June 1985, during a plant overhaul. An acceptance test performed by the customer right after the installation showed that the power consumption was 0.5 percent better than predicted.

Uprating of a Nitric Acid Plant

Plant Description

The original 600 tons per day nitric acid plant went in operation in 1981. The compressor set was never overhauled; it produced acid very reliably until 1986, when the installation was uprated. The plant is a mono-pressure unit, with oxydation and absorption at 7.0 bar before and 8.5 bar after uprating. Main data of the compressor train are tabulated as follows.

		old design	uprated design	start-up point
LP compressor				
Pressure ratio —		1.88	2.50	2.09
Flow	m^3/s	32.3	42.3	33.3
Power	kW	2,675	5,385	3,250
HP compressor				
Pressure ratio —		4,1	3,7	2,7
Flow	m^3/s	18,6	18,9	16,3
Power	kW	7,335	9,730	4,335
Expander				
Inlet pressure bar		6.08	7.1	4.35
Inlet temperature	°C	630	580	150
Power	kW	8,900	11,450	3,185
GEC-steam turbine				
Speed	rpm	6,700	6,750	5,600
Power required	kW	1,488	3,665	4,400
max.power available	kW	5,850	5,850	5,650

Plant Description

Axial Compressor AC 6-45 (LP). The LP compressor was the axial type (Figure 11). Original design was for six stages on a shaft of 450 mm diameter. This compressor has the bottleneck of the entire compressor set. Its relative air speed on the tip of first stage blades was already Mach 0.95. A further increase of the speed of rotation would, therefore, not lead to the corresponding flow increase. Since the original design used 80 percent of reaction, high Mach numbers were attained with a relatively low speed of rotation. Replacing this design 50 percent reaction blading reduced the Mach number from 0.95 to 0.70 for the same speed of rotation and flow. There is sufficient margin to increase flow by 33 percent (Mach 0.70 × 1.33 = Mach 0.93). The velocity triangle on Figure 12 explains this proposition.

A speed increase of almost 30 percent is, of course, not a practicable solution on an existing plant. Stresses in the existing machines would rise to unacceptable values. The most practicable solution for the present case was to increase shaft diameter from 0.45 m to 0.5 m (+ 11 percent). Applying the same tip to hub diameter ratio on the new shaft enlarged the flow channel by 23 percent. Only a small adjustment in rotational speed was, therefore, required to obtain the specified flow.

Boosting the LP compressor capacity was not the only task of the uprating. The discharge pressure was raised from 1.9 to 2.5 bar, in order to enter the HP compressor with about the same volumetric flow rate, as on its original duty. Aforemen-

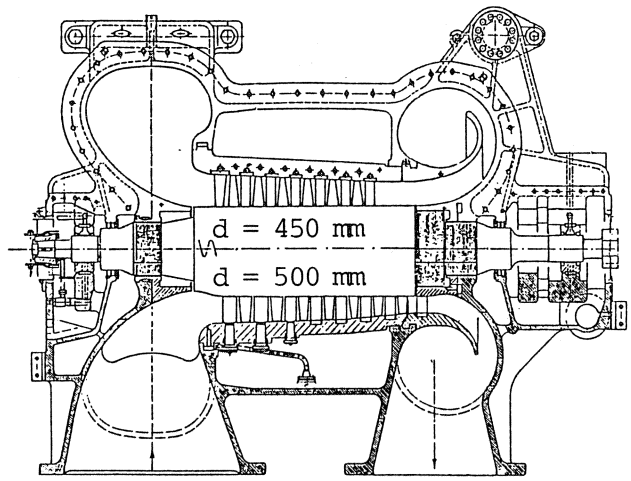

Figure 11. Axial Compressor Before and After Uprating.

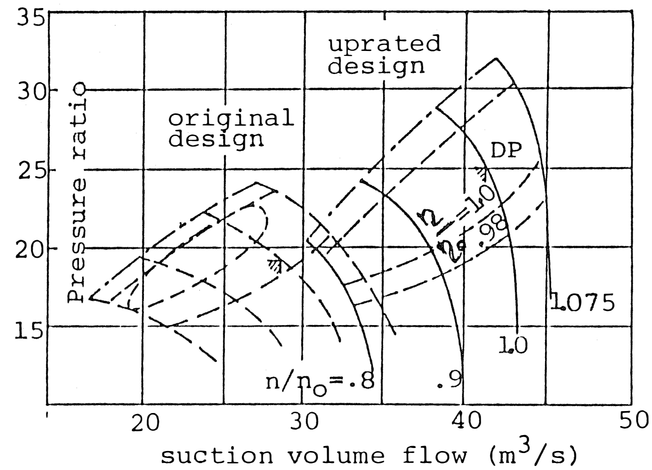

Figure 13. Comparison Between Original and Uprated Performance of Axial LP Compresor.

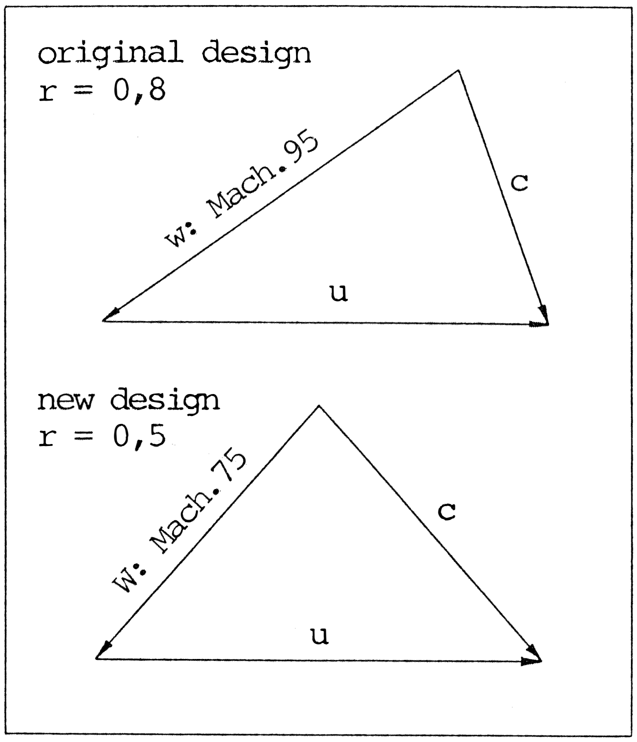

Figure 12. Velocity Triangles for Front Stages of Axial Compressor.

tioned increased shaft diameter lifted circumferential speed by 11 percent, sufficient to deliver the higher end pressure.

Rows one and two of the uprated compressor were finally designed for 50 percent reaction to cope with the increased flow, whereas rows three to seven still work with 80 percent reaction for a high discharge pressure. A seventh row was added solely to improve the surgeline at light off speed. Performance curves before and after uprating are shown on Figure 13.

Centrifugal Compressor R 71-.5 (HP). The flow increase and the increased discharge pressure of the LP compressor resulted in an unchanged suction volume flow to the HP compressor. Despite the power increase from 7,335 kW to 9,730 kW, no mechanical modification was required.

Tail Gas Expander AE 45-5. New inlet conditions for expander are calculated according to following equation:

$$\frac{\dot{m}\sqrt{RTZ_1}}{\sqrt{P_1^2-P_2^2}} = \text{constant} \tag{7}$$

Since the working pressure for oxydation and absorption was lifted by almost one bar, and the expander inlet temperature was lowered from 630°C to 580°C, this new duty increased the swallowing capacity of the expander by 20 percent. The expander had to bring the remaining 13 percent by increase of its flow area.

The swallowing capacity of this expander type is defined by the guide vane on Stage 1 (Figure 14). These blades are positioned by punched shims, which simply were exchanged by different ones. In fact, the expander in its original design was almost at its lowest capacity, while it is now used at its upper limit. No other parts were replaced nor modified.

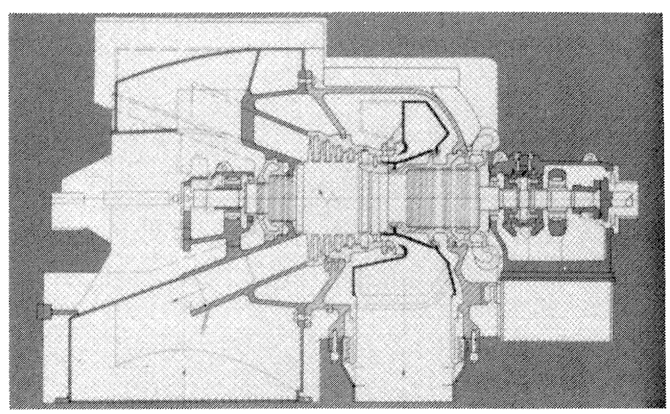

Figure 14. Tail Gas Expander.

Power recovered by the expander increased from 8,900 kW to 11,450 kW. Apart from this overall increase, a considerable change in power recovered of the individual stages and consequently, in bending stresses of the blades, was the consequence of this new duty, but there was sufficient design flexibility to allow for such variations.

Power Balance. Investigate whether or not the existing steam turbine provides sufficient power for the starting point. This starting point is at 5,600 rpm and at the highest possible discharge pressure for a clean stack. As can be seen from the listing of operating data, compressor power exceeds the available steam turbine power by far—7,585 kW, compared with 5,650 kW available on the turbine. In contrast to the original procedere, the expander inlet valve is now kept wide open for the starting of the uprated machine. The expander recovers 3,185 kW at an air inlet temperature of 150°C. This makes starting easy. Additionally, no switchover from the stack valve to the expander is required, once the process is started.

Miscellaneous. A complete review of all parts of such a compressor train is necessary when uprating it by one-third of its original capacity. This includes axial and journal bearings, control system, coolers, suction filters, silencers, etc. The decision about these parts, if they need to be replaced or modified, can determine whether or not the whole project will be interesting. In the present case, only a second blowoff line was required for the LP compressor.

Installation of Modified Parts

As outlined above, the modification work concentrated on an LP compressor and expander; steam turbine, HP compressor, coolers etc. remaining unchanged. The complete work was scheduled to be completed within eight days around the clock work. In the fact the compressor train was ready for trial runs after seven days and recommissioned in one and one-half days.

CONCLUSION

An analysis of the uprated plant demonstrated that the specified steam consumption, air flow, and swallowing capacity of the expander were met.

A simple exchange of rotor and stator in the air compressor, together with a bottlenecking of the plant, resulted in a production increase of 200 tons of nitric acid per day.

NOMENCLATURE

A	(m^2)	Area of flowchannel
D	(m)	Impeller diameter
c	(m/s)	Absolute flow velocity
h_p	(J/kg)	Polytropic head
$\dot{m}$	(kg/s)	Massflow
m	(−)	Polytropic temperature exponent
n	(−)	Polytropic exponent
Pp	(kw)	Gaspower/pressure
R	(J/kg deg)	Gasconstant
T	(°K)	Temperature
u	(m/s)	Circumferential speed
$\dot{V}$	(m^3/s)	Volume flow
W	(m/s)	Relative flow velocity
Z	(−)	Compressibility coefficient
η_p	(−)	Polytropic efficiency
μ_p	(−)	Polytropic head coefficient
μ_o	(−)	Polytropic work input coefficient
ν	(−)	Flow factor for axial stages $= \dot{V}_1/(u_h{*}A)$
ϕ	(−)	Flow factor for centrifugal stages $=\dot{V}_1/(u_2 D_2^2)$
π	(−)	Pressure ratio
ρ	(kg/m^3)	Density
δ	(deg)	Stator vane setting angle

Subscripts

1	Inlet of stage
2	Discharge of stage
p	Polytropic
h	Hub of rotor of axial compressor